SUBSTITUTE SPEFICATION



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TITLE: INTERCHANGEABLE 2-STROKE OR 4-STROKE HIGH TORQUE POWER ENGINE

This is a continuation-in-part of application no. 10/252,927 file date 09/24/2002 titled: Engine with 1-way clutch between a piston and power shaft.

## BACKGROUND OF THE INVENTION

Engines that transmit an offset piston's power to a straight power shaft have been attempted since at least 1921, e.g. patent no. 1,365,666 but have not had practical success though they inherently offer high torque and high fuel efficiency. Their weakness lies in using many energy absorbing moving parts and combustion chambers to convert the piston's reciprocating rectilinear motion to the power shaft's unidirectional rotary motion which has made them inefficient and impractical, e.g. patent nos. 2,239,663; and 5,673,665. For this reason, the simple, exhaust polluting, inefficient but reliable crank engine survives as the search for a better power source continues.

Enormous funds and research have been poured into fuel cells, electric vehicles and crank engine hybrids for years in an unsuccessful effort to replace the ubiquitous crank engine.

The crank engine is very inefficient because the two angles at both ends of the connecting rod of length **L** and the crank angle $\alpha$ (FIG 14) combine to slow the piston's speed, which traps the very rapidly expanding combustion gases in a small chamber. The gases build up very high heat and pressure at and near tdc. Here, nearly all the force from the pressure is vectored against the crankshaft's bearings instead of rotating it. Parts inertia is combined with extra fuel on each power stroke to overcome the angles' resistance. The result is excess exhaust pollution and waste heat. The waste heat is lost and the pollutants are partly scrubbed from the exhaust when it is too late.

The pollution and the waste heat must be reduced in the combustion chamber by converting them to mechanical motion with a more complete burn. To do that, all the rod and crank angles must be zero <u>during the entire power stroke</u> <u>but that is impossible</u> in a crank engine. The following mathematics explains why:

FIG 14 is a schematic that represents a crank engine. **FV1, FV2, FV3** are force vectors that come from burn pressure driving the piston **38**. **FV1** is along a radial of the crankshaft axis **C**. Only **FV3**, being tangent to the crank circle **d**, rotates the shaft where $\mathbf{FV3} = \mathbf{FV1}(\cos\theta)(\cos\Phi)$.

The crank engine's efficiency is zero at tdc when angle $\theta = 0^o$ but angle $\Phi = 90^o$, making **FV3 = FV1(1)(0) = 0**. When FV2 is tangent to circle d, $\text{Cos } \Phi = 1.0$ and $\text{Tan } \theta = r/L$ and $\theta = \text{Tan}^{-1} r/L$ from which $\text{Cos } \theta$ is found. The efficiency at that point is $FV3/FV1 = \text{Cos } \theta$. The importance of angle $\theta = \text{Tan}^{-1} r/L$ will be shown below.

The ratio of the displacement **M** along the crank circle **d** to the piston's displacement **a** at any chosen crank angle $\alpha$ is easily found from FIG 14. **r** is the crank arm length and $\alpha$ is in degrees:

$$r = b + a$$

$$a = r(1\text{-Cos } \alpha)$$

$$M = \pi\alpha r/180$$

$$M/a = \pi\alpha/[180(1\text{-Cos } \alpha)]$$

For instance, when $\alpha = 10^o$, $M/a = 11.49{:}1$. At this point, the rod's slow crank end must go **11.49** times as far as the piston. The slower the crank's rotation, the longer the gases are trapped in a small chamber and the lower the engine's efficiency. It is known that this is where the confined hot, pressurized gases create most of the pollution and waste heat. The crank's angular efficiency:

$$\text{Cos } \theta = FV2/FV1$$

$$\text{Cos } \Phi = FV3/FV2$$

$$FV2 = FV1(\text{Cos } \theta)$$

$$FV2 = FV3/\text{Cos } \Phi$$

$$FV3 = FV1(\text{Cos } \theta)(\text{Cos } \Phi)$$

$FV3/FV1 = (\text{Cos } \theta)(\text{Cos } \Phi)$ Crank engine's angular efficiency. It caps the burn efficiency.

FIG 14 is also the basis for the following indented equations that lead to the $\text{Cos } \theta$ and $\text{Cos } \Phi$ equations in terms of crank angle $\alpha$, length **L** and crank arm **r**:

$$180 - \beta = \gamma$$

$$\gamma + \theta + \Phi = 180$$

$\beta = 90 - \alpha$ Note the rt. triangle $(\alpha + \beta + 90)$

$180 - (90 - \alpha) = \gamma$ or $90 + \alpha = \gamma$

$$(90 + \alpha) + \theta + \Phi = 180$$

$$\alpha + \theta + \Phi = 90$$

$$n = r \sin \alpha$$

$$\sin \theta = (r/L)\sin \alpha$$

$$\theta = \sin^{-1}[(r/L)\sin \alpha]$$

$$\cos \theta = \cos\{\sin^{-1}[(r/L)\sin \alpha]\}$$

$$\alpha + \sin^{-1}[(r/L)\sin \alpha] + \Phi = 90$$

$$\Phi = 90 - \{\alpha + \sin^{-1}[(r/L)\sin \alpha]\}$$

$$\cos \Phi = \cos(90 - \{\alpha + \sin^{-1}[(r/L)\sin \alpha]\})$$

The equations **Cos θ, Cos Φ** are easily solved with a hand calculator. For instance, they give the *angular efficiency* = **22.4%** when **α = 10°; r = 1.5 "; L = 5.0 "**. Since the *burn efficiency* is low (See **M/a** above) the *total efficiency* has to be much less than **22.4%** in this example. The efficiency increases as **α** increases but the combustion pressure decreases as **α** increases. A higher rpm increases efficiency but that has reached its limit and it is not good enough.

The importance of angle **θ = Tan⁻¹r/L** now follows. That is when **FV2** is tangent to the circle **d** at the arm **r** which makes angle **Φ = 0.0** and **Cos Φ = 1.0**. The *angular efficiency* is **Cos θ = Cos(Tan⁻¹r/L).** In the example above where **r = 1.5 "; L = 5.0 "; FV3/FV1 = Cos θ = 95.8%.** Extend **L** relative to **r** so that angle **θ** goes to **0.0.** Then $\lim_{\theta \to 0.0} \cos \theta = 1.0$. (This is the foundation for differential calculus). That makes the *angular efficiency* **FV3/FV1 = (Cos θ)(Cos Φ) = (1)(1) = 100%** because there is no angular resistance since the angles **θ,Φ** disappear. The variable angle **α** disappears. The crank arm **r** disappears. The variable length torque arm **n** (FIG 14) which requires torque buildup is replaced by the fixed length torque arm **r'** (FIG 15) which gives instant peak torque.

Unlike the crank, **FV1** in this invention (FIG 15) is always directed to rotating the output shaft **8** rather than directed against the shaft's bearings. **FV1** is transmitted with both angles **θ,Φ = 0.0** through the entire power stroke. The **M/a = 1:1** through the entire stroke. The circumference **d'** replaces the crank circle **d** in FIG 14. Motion is transmitted through the fixed length torque arm **r'** to the output shaft 8.

BRIEF SUMMARY OF THE INVENTION

This is a high torque power, fuel-efficient engine that can be easily switched between a 2-stroke and a 4-stroke. A pair of combustion cylinders and their related pairs of parts, including 1-way

clutches, are connected by an idler gear to make the basic 2-stroke engine. A third idler connects two pairs to make a 4-stroke engine. Computer controlled ignition allows power stroke overlap by equally spaced-apart pistons. The crankshaft is replaced by a straight power shaft.

A rugged 1-way clutch transmits motion between the power piston and the output shaft. The piston is offset from the shaft's axis by the radius of the 1-way clutch at the point where it engages the piston connecting rod. Though conventional 1-way clutches will work, they are inefficient because they transmit motion between the races through two vectors. One vector is parallel to the clutch radial, which does not transmit motion. Instead, its energy is converted to waste heat that can contribute to early clutch failure. A preferred 1-way clutch that efficiently transmits torque between its races perpendicular to a clutch radial is described below with reference to FIGs 7-13.

The math below can be used to calculate important values in designing a 2-stroke and a 4-stroke.

Objects of this invention include:

1. easily interchanged between 2-stroke and 4-stroke;
2. low cylinder expansion rate with a small bore, which allows more complete combustion of a small combustion charge resulting in high fuel efficiency;
3. instant peak torque at the beginning of the power stroke;
4. the 1-way clutch overrun feature allows deactivating pairs of pistons without load on the shaft;
5. reduced mass engine compared to a crank engine;
6. a rugged breakaway 1-way clutch that is easily disassembled and reassembled for repairs;
7. lightweight piston and rod due to compression forces only;
8. piston always square in its cylinder reduces cylinder wear;
9. power stroke overlap.

BRIEF DESCRIPTION OF THE DRAWINGS

In the drawings: FIGs 2,3 show a representative 1-way clutch of any suitable design but a preferred rugged design in which motion is transmitted between races perpendicular to clutch radials is described with reference to FIGs 7-13. Number 89 refers to a cover plate in FIGs 10,13 and to a cover plate with cartridge, including its elements in FIGs 7,8. The outer race is referred to by its separate parts 5A, 5B and 5C in FIGs 7,8 and as a whole by the number 5 in the other FIGs. Number 82 and number 96 in FIGs 7,8 refer to equivalent parts. The output shaft is represented by its axis 91 in FIG 8. Parts are shown with solid lines in drive and dashed lines in overrun.

FIG 1 is a side view showing how movement of parts is synchronized between a pair of pistons.

FIG 2 is taken essentially along line 2-2 in FIG 1 to show how motion is transmitted between a piston and a 1-way clutch through a gear mesh.

FIG 3 shows how a belt or a chain replaces the gear mesh in FIG 2.

FIG 4 shows a means for decelerating and reversing pistons at the end of the stroke.

FIG 5 shows two computer controlled pairs of cylinders combined with an energy storage device.

FIG 6 shows a 4-stroke engine by combining two pairs with a third idler 40A.

FIG 6A focuses on separation of idler 40A from the sector gears in FIG 6 to create a 2- Stroke.

FIG 7 shows an oblique view of the 1-way clutch with keystone shaped interlocking teeth on the outer race.

FIG 8 is an exploded view of the several parts of the FIG 7 clutch aligned along a shaft axis. Alternatively, pegs with matching holes replace the teeth in FIG 7.

FIG 9 is a side view of a replaceable clutch cartridge with its cover plate removed and casing broken away to show the internal elements of a hydraulic motion transmitting member.

FIG 10 is a cross sectional along 10–10 in FIG 9.

FIG 11 is one embodiment of a mechanical transmitting member.

FIG 12 is a second mechanical embodiment of a transmitting member.

FIG 13 shows a cross sectional along 13-13 in FIG 11.

FIG 14 is a schematic of a crank engine used for mathematical reference in the text above.

FIG 15 is a schematic of this invention used to mathematically compare with FIG 14.

## DETAILED DESCRIPTION OF THE INVENTION

First, consider the benefit of overlapping power pistons on the power stroke e.g., a 2-stroke, 6 cyl engine with a 9" piston stroke would simultaneously have the $1^{st}$ piston 6" after tdc, the $2^{nd}$ piston 3" after tdc and the $3^{rd}$ piston igniting at tdc. The 6 pistons continuously cycle through their power strokes in this sequence. The power added by the $3^{rd}$ piston is reduced by the combined remaining power of the $1^{st}$ and $2^{nd}$ pistons resulting in fuel savings and smooth power shaft rotation.

**Underlying Mathematics.**

Definitions:

**1 BTU** = 778 ft-lbf

**1 hp** = 550 ft-lbf/sec.

**2πr′** = length of 1-way clutch rim at connecting rod contact. (ft)

**bore** – cylinder diameter. (in.)

**Cp** – cylinder pressure calculated from known bore size. (psi)

**Dp** – displacement (cu.in.)

**E** – fuel efficiency

**F** – combustion force per piston. (lbf)

**Fg** – fuel flow rate (gals/hr)

**Fi** – force on the inner race (lbf)

**Fr** – fuel flow rate (lbm/sec)

**Fu** – force per unit 89 (lbf) See FIG 7 or FIG 8 for unit 89.

**Fw** – fuel's weight (lbm/gal.)

**hp** – shaft horsepower.

**k** = 2 or 4 (k = 2 for a 2-stroke. k = 4 for a 4-stroke.)

**Lo** – fraction of power lost

**n** – number of active pistons. 2,4,6,8, …

**n/k** – number of overlapping pistons cycling through the power stroke.

**Nu** – number of units 89 (FIGs 7,8).

**Pp** – estimated combustion pressure per piston. (psi) Used to find the bore size. (in.)

**Ps** – length of piston's stroke. (in.)

**Qc** – fuel's energy density. (BTU/lbm)

**r** – radius of cylinder. (in)

**r′** – 1-way clutch radius at connecting rod contact. (ft)

**ri** – radius of the 1-way clutch inner race. (ft)

**Rv** – power shaft's rotation rate. (rpm)

**Sp** – Center to center spacing between units 89 (FIGs 7,8). (ft)

**T** – torque per piston. (lbf-ft)

**T′** – total shaft torque. (lbf-ft)

**Vp** – piston velocity. (ft/sec)

Equations:

**Vp = π(r′)(Rv)/(30)** Piston rod's and the 1-way clutch's rim speeds are equal at contact.

**r′ = 30(Vp)/π(Rv)** **r′,Vp, Rv** are central to this engine's design and operation.

**Rv = 30(Vp)/(πr′)**

**F = 550hp(k)/(nVp)**

**hp = F(n)(Vp)/550**

**hp = Fr[778(Qc)(1-Lo)]/550**

**T = F(r′)**

**T′ = nT/k**

**Pp = F/[π(r²)]**

$r^2 = F/(\pi Pp)$

$bore = 2[F/(\pi Pp)]^{.5}$

$F = \pi(Pp)(bore^2)/4$

$Fi = F(r')/ri$

$Nu = 2\pi(ri)/Sp$

$Fu = F(r')(Sp)/[2\pi(ri^2)]$

$Fu = F(r')/[(ri)(Nu)]$

$Fu = Fi/Nu$

$Cp = 4F/(\pi bore^2)$

$Dp = \pi(bore/2)^2(Ps)(n)$

$Fr = 550hp/[778(Qc)(1\text{-}Lo)]$

$Lo = 1 - 550hp/778(Qc)Fr$

$E = 1 - Lo$

$E = 550hp/778(Qc)Fr$

$Fg = Fr(3600)/(Fw)$

The following example demonstrates the effectiveness of the Underlying Mathematics in finding the correct general engine specifications from which the rest of the engine can be built. The given values are hypothetical. This example is for a low power engine, e.g. lawn mowers and outboard marine, but the math can be applied to any size engine.

Example:

Given: **Pp** = 100 psi; **F** = 300 lbf; **Vp** = 3.5 ft/sec; **r′** = 4.5" = .375 ft; **ri** = 3.75" = .3125 ft; **k** = 2; **n** = 2; **Qc** = 20500; **Lo** = .35; **Fw** = 6 lbm/gal; **Sp** = 6" = .5ft

**hp** = 300(2)(3.5)/[2(550)] = **1.909**

$r^2$ = 300/(100π) = **.9549** in$^2$

**bore** = 2[300/(100π)]$^{.5}$ = **1.9544** in.

**Rv** = 30(3.5)/(.375π) = **89.13** rpm

**Fi** = 300(4.5)/3.8 = **355.37** lbf

**Nu** = 2π(3.8)/6 = **4**

**Fu** = 300(4.5)/[6(3.75)] = **60** lbf.

**T** = 300(.375) = **112.5** lbf-ft

**Fr** = 550(1.909)/[778(20500)(1-.35)] = **.000101284** lbm/sec.

**Fg** = .000101284(3600)/6 = **.060770629** gals/hr.

Given: **hp** = 10; **F** = 380 lbf.

**Vp** = 550(10)(2)/(2)(380) = **14.5** ft/sec.

**Rv** = 30(14.5)/(.375$\pi$) = **369** rpm.

**Fi** = 380(4.5)/3.8 = **450** lbf

**Fu** = 380(4.5)/[6(3.75)] = **113** lbf.

**Cp** = 4(380)/[($1.9544^2$)$\pi$] = **126.67** psi.

**T** = 380(.375) = **142.5** lbf-ft

**Fr** = 550(10)/[778(20500)(1-.35)] = **.000530537** lbm/sec.

**Fg** = .000530537(3600)/6 = **.318322345** gals/hr.

**Discussion.**

A pair of combustion cylinders 33 and related pairs of parts that include a pair of 1-way clutches (FIGs 1-3) make the basic 2-stroke engine in this invention. The clutch's inner race 4 is keyed to the power shaft 8. The outer race 5 carries a sector gear 12. Each gear 12 engages an opposite side of idler 40 whereby synchronous reverse motion is transmitted between the power piston 38 and the second piston 38 in the pair as the inner race 4 transmits the power to the shaft 8. Moving parts that are not shown with arrows 42 are presumed obvious.

Combining two pairs with idler 40A creates a 4-stroke shown in FIG 6 that will be described later under Interchanging 4-stroke and 2-stroke.

FIG 2 shows a gear mesh to transmit the piston's power between piston rod 18 and the outer race 5 of the 1-way clutch. Rod 18 reciprocates along a straight path 42. FIG 2 also shows a reciprocating starter 46 gear meshed with the outer race 5. By shifting race 5, the starter shifts both pistons 38 until ignition. Alternatively, shaft 43 can be used to shift the pistons until ignition. The 4-stroke version in FIG 6 needs one starter 46 (not shown).

One end of a V-belt or a chain 9 is fastened to the outer race 5 (FIGs 1,3). The way it is wrapped around race 5 always keeps it taut, which prevents backlash as it rotates race 5 in response to the power stroke. Rod 18 is connected to the other end of the belt or chain 9 with a suitable fastener 41.

The 1-way clutch's override feature in this engine allows output shaft 8 and the clutch's inner race 4 to rotate independently of the pistons 38 when the inner race's speed is greater than the outer race 5 speed. This feature creates regenerated energy is collection in an energy storage device 26 (FIG 5) available, e.g. for dumping to shaft 8 on demand or generating electricity.

The fixed length torque arm 10 (FIGs 2,3) causes instant peak torque at the beginning of the

power stroke. A connecting rod guide 21, secured to housing 15, eliminates side thrust and reduces wear by keeping the piston 38 square in its cylinder. Wrist pins and piston skirts are not needed. The guide 21 is combined with a decelerator mechanism (FIG 4) to stop piston 38 at or near top dead center. The decelerator includes a node 19 that is part of each rod 18 in a pair and a spring 45 for each node. The spring is encased in the guide 21. An opening in the housing 15 allows easy replacement of the spring. The spring absorbs the impact of node 19 to halt the motion of piston 38, which is then accelerated on its power stroke by timely expanding combustion gases. The impact is reduced because node 19 is decelerating due to the power loss of the power piston to the shaft 8. The decelerator is positioned to prevent backlash of the gears 12 (FIGs 1,6) that mesh with idler 40.

A computer 7 (FIG 5) monitors input from the throttle 6 and shaft power from the sensor 22 on shaft 8 through leads 23 to determine the size of the combustion charge to transmit to the cylinders through injector lines 24. The position of piston 38 is monitored through sensors 22 on shaft 43 and used for ignition timing. By monitoring the motion of each shaft 43 in several pairs, the computer controls timing between the unconnected pairs in a 2-stroke embodiment. The computer begins a power stroke with a piston in one pair when a piston in another pair is partly through its power stroke. In a 2-stroke, 50% power stroke overlap and smooth rotation of the shaft 8 is had with two unconnected pairs (four cylinders). Greater overlap is gained with more pairs.

**Moderated Combustion Pressure.**

There are least three ways to absorb excessive peak cylinder pressure and dispense it back to the chamber 33 so that a moderated pressure is maintained during the piston stroke to achieve a better burn which increases efficiency and reduces pollution and waste heat.

The first way (FIG 4) uses a two-part piston rod 18 and 18A with a spring 16 between the two parts. Spring 16 is connected to the two parts such that its compression and expansion are not affected. Part 18 has an extension 14 that extends through the center of spring 16 into a cylinder 13 in part 18A (shown in cross section) to keep the spring 16 centered on the axis of the two piston rod parts. Side thrust, if any, will be negligible on the parts because of the combined guide 21 and the piston 38 being square in the cylinder 33.

There are two channels 2 on opposite sides of the cylinder 13 that are aligned with the axis of the cylinder. A small projection 3 on the extension 14 reaches into each channel to prevent angular motion of part 18A and piston 38.

A second way includes a small, suitable flywheel 48 splined to the end of shaft 43 (FIG 3). A conventional flywheel can be used but an alternative comprises three concentric parts. The inner part is splined to shaft 43. The outer part extends to the flywheel's rim. Between them is a tough, slightly elastic part that absorbs some of the initial ignition jolt.

A third way is to construct the inner race 4 with springs like the flywheel carried behind the engine of conventional vehicles. The inner race performs like the flywheel.

**Interchanging 4-Stroke and 2-Stroke.**

With reference first to FIG 3, the flexible chain 9 must be made stiff enough to pull the piston 38 down during the intake stroke in the 4-stroke engine. In this case, the outer race 5 is a cogwheel and the cogs fit between the chain links similar to a bicycle chain. Each side of each link has an extension that rides in an immovable channel that is secured to the engine. The two channels combine with the side extensions to prevent the chain from flexing out of mesh with the race 5 cogwheel during the intake stroke and without interfering with the other piston strokes. Both channels are shaped in an arc around the race 5 where they are connected with a solid cover over the chain to insure against the chain flexing. The cover ends near the position of fastener 41 in FIG 3 when piston 38 is at top dead center. The channels continue straight downward without the cover to prevent the chain from flexing when it is straight. The straight channels extend to a point slightly beyond the position of fastener 41 when the piston is at bottom dead center. The fastener is connected to the chain free of the extensions so that the channels do not interfere with the motion of the chain. This allows the fastener 41 to reach its highest point (FIG 3) where it is in position to begin the intake stroke and complete the stroke without interference from the channels or the cover.

There are at least two simple ways to change between a 2-stroke and a 4-stroke. In a 4-stroke, a sector gear 12 on two pairs engages idler 40A (FIG 6). A removable cap 54 having a hole is threaded to the engine 15. The shaft 43 of idler 40A has two diameters. The shorter one extends through the hole. A snap ring 56 on the shorter diameter abuts the cap and combines with the larger diameter that abuts the inside of the cap to prevent the idler 40A from axial movement which keeps the idler properly engaged with the two sector gears. When changing to a 4-stroke from a 2-stroke, the pistons must be correctly positioned before engaging the idler with the sector gears. One of the correct positions is shown in FIG 6 with 2 pistons at top dead center and 2 at bottom dead center. Power stroke overlap for a 4-stroke can be achieved by adding another bank of two pairs along the shaft 8 disengaged from the bank shown in FIG 6 or by adding separate pairs.

The separation 1 in FIG 6A makes the 4-stroke a 2-stroke. To change to a 2-stroke from a 4-stroke, the cap 54 is partly unscrewed to a predetermined position on the engine 15, which raises shaft 43 and disengages idler 40A from sector gears 12 (FIG 6A). The cap is held in place by known means, e.g. a dowel through the side of the cap that contacts engine 15.

**Hydrogen Enhanced Ignition.**

In some applications, considerable regenerated energy from shaft 8 is anticipated from the 1-way clutch's overrun feature. The device 26 (FIG 5) includes a means (not shown) to convert the energy to hydrogen ($H_2$) and a temporary $H_2$ storage tank. A minimum of the $H_2$ is injected into the combustion chamber with the primary fuel.

Hydrogen's "flame speed" in an $H_2$ rich mixture is about 6 times faster than gasoline. (Energy Technology HDBK, pp. 4-39 to 4-43, Considine, 1977). $H_2$ has a high energy density in the high

pressure combustion chamber. High heat from the ignited $H_2$ saturates the primary fuel to cause a more complete burn of the primary fuel's droplets, which increases fuel efficiency. The high, prolonged pressures that cause NOx will be greatly reduced if **Vp** and **r'** are selected to allow a fast piston acceleration to reduce the pressure. If needed, flywheel 48 fine adjusts the acceleration and pressure for the best burn. **M/a = 1:1** (See **M/a** above) and the angles **$\theta$,$\Phi$,$\alpha$** (FIG 14) do not exist.

**<u>Parabolic Reflector Cylinder Head</u>.**

A drawing is believed not necessary to describe this embodiment. The entire cylinder head is a parabolic reflector with an igniter at its focus. The focus is at the end of a replaceable plug. An energy wave expands from the igniter to hit the parabolic reflector and the reflector directs the energy wave to uniformly impact the flat piston crown at or near top dead center. Both pistons in a pair will be decelerating due to power bleed and the additional wave energy will help to reverse and accelerate both pistons 38 from zero where it is most effective in saving fuel.

**<u>1-Way Clutch with Axial Extension</u>.**

This embodiment is also not shown with a drawing. In FIGs 1-3, the rod 18 engages the outer race 5 at its rim. If the rim radius cannot be reduced enough to obtain sufficient combustion pressure, an extension of race 5 along shaft 8 has a shorter radius. The rod 18 engages the extension's rim at the shorter radius rather than the race 5 rim. Rod 18 reciprocates along its straight path, tangent to the extension's rim. Motion from combustion pressure is transmitted to the race 5 extension. Race 5 transmits the motion to the inner race 4 at the longer radius.

**<u>Preferred 1-Way Clutch Embodiment</u>.**

The preferred breakaway 1-way clutch is shown in FIGs 7-13. Its outer race 5 drives clockwise in its indexing motion 42. The outer race 5 has three separate parts: sides 5A, 5C and race 5B. Race 5B is the outer rim of the gap 28 (FIGs 7,9-11,13). The gap is narrow and near the race 5B to reduce stress on the parts. FIG 7 shows the torque transmitting units 89 in relation to the gap.

Keystone shaped teeth 82 (FIG 7) extend from race 5B and make a strong interlocking fit with keystone shaped teeth 96 on the sides 5A and 5 C The fit locks the parts together radially and circumferentially but allows them to be easily moved axially for disassembly by removing the snap rings 90 (FIG 8). FIG 8 shows equivalent pegs 82 that fit into holes 96 in sides 5A and 5C. There are as many teeth or equivalent pegs as needed.

The inner race 4 is keyed to power shaft 8. A snap ring 90 carried by shaft 8 on each side of the race 5 (FIG 8) keeps the clutch from shifting along the axis 91 of shaft 8. The snap rings also prevent separation of the three outer race parts. In extreme or unusual use, a dowel 17 (FIGs 7) reinforces the

snap rings to keep the parts together. It extends through race 5A and 5C to contact a keystone shaped tooth 82 (or an equivalent peg 82 in FIG 8) on each side of race 5B. It is easily displaced for breakaway to replace race 5B.

FIGs 7,8 show two halves of race 5B that are kept in contact 94 by the teeth (or pegs). When race 5B is separated from sides 5A and 5C, the halves fall apart for replacement without separating the other parts from shaft 8.

Bearings in FIG 7 are between the outer race 5 and the shaft 8. Spokes 35 in side 5A and side 5C reduce material cost and reduce indexing inertia. The transmitting units 89 are easily replaceable when positioned between the spokes or behind an aperture in the sides 5A and 5C.

Move the bearings to the conventional position at gap 28 and the dowel (FIG 7) can keep the parts together without the spokes 35.

The cover plate 89 (FIGs 12,15) is designed to guide the moving parts during their movements.

**Hydraulic Embodiment of the 1-Way Clutch.**

Replaceable hydraulic cartridges 89 (FIGs 7,8) are carried by race 4. The race is molded to rigidly hold the cartridge casing 80. Pegs 92 (FIG 9) slide into grooves in the race 4 to reinforce the cartridge against movement, especially toward race 5 under centrifugal force. A unit piston 81, shown in driving contact with race 5 (FIGs 9,10), moves a short distance 88 along the clutch radial 93 (FIG 9) while in sliding contact with the casing 80 and the casing is in contact with race 4. The piston is secured to a piston rod 84 (FIGs 9,10) that is hydraulically actuated from a reservoir section of the casing from which it extends. Torque between race 5 and race 4 is transmitted through the piston perpendicular to radial 93 that extends from the axis 91 (FIG 8) of shaft 8. The casing 80 has an arm that holds a plunger 79 in contact with the ball end of a trigger 85. A cap 86 having a slot aligned with the trigger's motion is immovably secured to the arm. The trigger extends through the slot to contact the race 5. A resilient piece inside the cap between it and the ball end is preferred. The angle between the arm and the radial is small to prevent jamming between the arm and the trigger.

As the trigger 85 shifts from its overrun position to the drive position, it pushes the plunger 79 farther into its arm to displace hydraulic fluid in the reservoir contained in the casing 80. The fluid displaces the piston rod 84 to drive the piston 81 into non-slip contact with race 5. The piston is in contact with race 4 and drive is transmitted from race 5 through the piston to race 4 perpendicular to a clutch radial. One contact surface of the piston or race 5 should have a V-groove and the other shaped to increase non-slip friction upon contact. The trigger's motion is unhindered as it moves the

piston from the overrun position 88 to contact the race 5, except for compressing a resilient element 83 (FIGs 9,10).

The two-part resilient element 83 fits around the rod 84 for easy replacement. The element is positioned between a plate 87 that is part of the rod and a two-part, immovable second plate 60 that is part of the casing 80 and cover plate 89. When the trigger shifts to its drive position, the element is compressed between the two plates as the hydraulic fluid drives the rod 84 to bring the piston and race 5 into non-slip contact. The element expands against the immovable plate 60 to shift the piston to its overrun position 88 when the trigger shifts to its overrun position and releases the fluid pressure.

**Mechanical Embodiments of the 1-Way Clutch.**

Two of at least three mechanical versions of the transmitting units are shown in FIGs 11,12. A casing for them is omitted to show a cost saving but can be included. The cover plate 89 and race 4 substitute for the casing 80. Without a casing, the piston 81 is always in direct, sliding contact with race 4 as it reciprocates along the radial 93 that extends from the clutch axis 91 (FIG 8). Like the hydraulic version, the short reciprocal motion goes between contact with the race 5 and position 88. Drive is transmitted perpendicular to the radial 93 from race 5 through the piston to race 4.

FIG 11 shows the piston connected to a piston rod 101 by a wrist pin 97. The rod is connected to a lever 100 which, in turn, is connected to the trigger 85. All the connections are hinged to allow pivoting. The lever's fulcrum 99 extends from race 4. A cantilevered fulcrum (not shown) uses a snap ring or common washer and cotter pin to retain the lever. But a stronger fulcrum fits into a hole in the plate 89 (FIG 13) which is preferred for heavy duty. Three pegs 30, placed at the apexes of a broad triangle on plate 89, rigidly fix the plate to the race 4 in all embodiments. The angle between the lever 100 and the trigger 85 equals or is very close to 90° in the drive position to reduce stress on the trigger and its connection with the lever. The angle between the rod 101 and lever is preferably not straight when the piston contacts race 5. After contact, the angle straightens to increase pressure between the piston, the race 5 and lever's fulcrum 99 with limited force upon the trigger. A spring 11 insures instant separation of the piston 81 from race 5 as overrun begins.

The second mechanical version is shown in FIG 12. Some reference numbers for the same parts in FIG 11 are omitted in FIG 12 to avoid overcrowding. A lever 100 oscillates on its fulcrum 99 which extends from race 4. A gear mesh combines lever 100 with rod 84 to shift piston 81 into and out of contact with surface 112 on race 5. The piston is shown in contact with surface 112. The single piece rod 84 and piston 81 shift along a clutch radial 93 while in sliding contact with the carrying

race 4. Space 88 allows the shift. Only a few teeth complete the gear mesh since the rod's motion is very short. A very short motion reduces backlash and may even make it negligible. If short enough, the gear mesh could be eliminated in favor of a single piece lever and rod. The spring-loaded trigger 85 at the end of arm 3 extends across gap 28 and stays in contact with the tough, long wearing strip 14 carried by race 5. The piston never contacts the strip 14. The trigger slides over strip 14 during overrun and grabs it at the beginning of the power stroke to oscillate the lever in response to the motion of race 5, thereby shifting the rod and piston. Torque is thus efficiently transmitted to race 4 perpendicular to the clutch radial 93.

Not shown is a third mechanical version that sets the piston on one radial of the clutch and the fulcrum on another. It can also eliminate the rod 101.

In all the 1-way clutch embodiments shown in FIGs 9-11,13: (1) the angle at the trigger's two extreme positions must not cause jamming, (2) the trigger should be suitably coated and shaped to reduce drag but instantly grab the outer race when reversing to the drive direction, (3) the piston's motion 88 goes only far enough to provide clearance between the piston and the outer race during overrun and (4) one of the contact surfaces has a common V-groove and the other contact surface is beveled to fit it to prevent slip.